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RESEARCH ARTICLE - MECHANICAL ENGINEERING

A Study of Operating Characteristics of Old-Generation Diesel Engines Retrofitted with Turbochargers

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Abstract In developing countries, a large number of old-generation diesel engines without turbochargers have become popular, and they are the main sources of power in the area of transportation. However, these types of engines have many disadvantages, such as low power and efficiency as well as a large amount of exhaust emissions. Meanwhile, these countries cannot immediately cease their image for them due to the limitations of budge for industry development. In this research, we promote a method to improve the efficiency of operation of old model diesel engines used for agricultural machines in Vietnam by retrofitting turbochargers. Consequently, the operating characteristics of retrofitted turbocharger engines have been performed both experimentally and using simulation. The research results show that the engine performance and exhaust emission improve dramatically when retrofitting suitable turbochargers corresponding with the engines. However, after turbocharging, we also increase the thermal and mechanical stresses of engines. The results indicate that we need to perform in detail to find the best solution for the stable operation of engines.

Keywords Internal combustion engine · Pollution emissions · Turbocharger · Compressor · Volumetric efficiency

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1 Introduction

Diesel engines, which are the primary power source of light-duty, heavy-duty vehicles and agricultural machines, contribute importantly to the industrial development of Vietnam. In 2013, approximately 0.8 million light-duty and heavy-duty vehicles were registered in Vietnam. Studies showed that Vietnam's vehicle fleet has grown consistently at an average of 16% per annum, and growth in cars is highest at 18% during the same period [1]. However, a large number of old-generation diesel engines without turbocharge remain popular throughout the country. After more than two decades of operation, these engines have been degraded and are a primary cause of pollution. In addition, it is very difficult for a developing country as Vietnam to renew all types of engines owing to limited funds. In order to improve the air quality, the Ministry of Transport of Vietnam has declared the process of tightening the emissions standards for imported and manufactured vehicles, which corresponds to the move to Euro 4 emissions standards for new vehicles starting July 1, 2017 [2]. Nevertheless, Vietnam as well as other developing countries cannot immediately eliminate old model vehicles retrofitted with non-turbocharger diesel engines. These problems will continue to increase the state of pollution in the environments of these countries. As a result, they have to find the best solutions to adapt for both industrial development and pollution prevention.

To solve this problem, in this study, we propose a solution to enhance the power density of these engine by retrofitting turbochargers and optimizing the designs according to each type of engine. Exhaust gas from engines has high pressure and temperature, which accounts for approximately 30% of the total energy input [3]. There are many studies related to exhaust heat-recovery methods such as turbo-compounding, Rankine cycle, thermo-electrics, thermo-chemical recupera-

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tion, in-cylinder waste heat recovery [4–8] and turbocharging is an effective method. By turbocharging, not only can the outputs of internal combustion engines (ICE) be increased to match the output of substantially larger ICEs, but there is a reduction in some emission components [9–16]. Superchargers, where the engines themselves power the impeller via a short drive shaft, were originally known as the first method for turbocharging engines. However, turbochargers have become more and more popular in several decades because they have many advantages [17–21]. The important key difference between a conventional supercharger and a turbocharger is that the former is connected to the crankshaft and directly driven by the engine. On the other hand, a turbocharger is connected to a turbine that is powered by using the waste energy from the engine's exhaust gas; consequently, turbochargers tend to be more efficient and are commonly used on heavy truck, trains, as well as agricultural and construction vehicles that utilize diesel engines. The effects of the turbocharger and fuel on the performance and exhaust emissions of diesel engines have been described in [22]. The application of a turbocharger to the engine is responsible for the increases in the brake power and torque with both biodiesel and diesel fuel. However, the rate of increase in the brake power and torque with biodiesel is higher than with diesel fuel for turbocharged operation. In another study, ArtoSarvi et al. showed that the operation mode of a diesel engine significantly affects the exhaust emissions. Consequently, the exhaust emission was also largely dependent on the engine turbocharger system, in particular the bypass control [23].

In this study, we present a simulation and experiment to evaluate performance parameters and exhaust emissions of an agricultural diesel engine for the cases of both naturally aspirated and turbocharged conditions. We performed the simulation and experimental tests for various engine speeds and full-load conditions using the AVL Boost software and engine dynamometer test bed. Consequently, for comparison, we determined the performance characteristics as well as the volumetric efficiency, in-cylinder temperature and pressure.

2 Experimental Procedure

2.1 Test Engine and Fuel

The test engine used was originally a four cylinder, non-turbocharger diesel engine with a power output of about 60 kW. The details of the engine specifications are given in Table 1. The fuel fired was diesel fuel, which is commonly sold in the Vietnam market with properties as given in Table 2.

Table 1 Engine specifications

No	Parameter	Value	Unit
1	Cycle	4-stroke DI	–
2	Firing order	1-3-4-2	–
3	Displacement volume (V_h)	4.75	dm^3
4	Bore/stroke (D/S)	110/125	mm/mm
5	Connecting rod length	237	mm
6	Pressure ratio (ε)	17	–
7	Rated power	60	kW
8	Maximum torque	280	N m
9	Injection pressure	22	MPa
10	Timing injection (φ_s)	12	°BTDC

2.2 Turbocharger Selection for the Test Engine

The selection of a suitable turbocharger for a naturally aspirated diesel engine should consider not only the facts related to the specific engine required, but also the operating reliability. One of the most important factors in determining which turbocharger is most appropriate is to have a target power for the application. However, we should also have a realistic mindset because we need to consider the additional collection of modifications. The original power of the test engine with a cylinder volume of 4.75 l is about 60 kW and its parameters are highlighted in bold (see Table 3). This means that the power density of the test engine is only 12.3 kW/l of cylinder volume, which is significantly smaller compared with other diesel engines used in Vietnam, as shown in Table 3.

In this study, we aimed to increase the power of the test engine to 110 kW by retrofitting a turbocharger for the engine. As a result, its maximum power density increased from 12.3 to 22.55 kW/l of cylinder volume. The power of a non-turbocharger or turbocharger internal combustion engine is described in Eq. 1 [24]:

$$N = V_h.\eta_v.\rho.\frac{Q_H}{M}.\frac{n}{30\tau}.\eta_m.\frac{\eta_i}{\alpha}.i \tag{1}$$

where N is the power of a non-turbocharger or turbocharger internal combustion engine, V_h is the displacement volume of the piston, η_v is the volumetric efficiency, ρ is the air density of non-turbocharger or turbocharger engine, and Q_H is the heating value of fuel. M is the theoretical mass of air required to completely burn 1 kg of fuel, n is the engine speed, τ is the engine stroke, η_m is the mechanical efficiency, η_i is the indicated efficiency, i is the cylinder number, and α is the air excess ratio of fuel.

Assuming that some parameters, including $V_h, \eta_m, \eta_i, Q_H, n, \tau, \alpha, I$ and η_v, are constant because the changes in these parameters are trivial in comparison with the change in

Table 2 Diesel fuel specifications used in this research

No	Fuel property	Value	Test method
1	Sulphur content (mg/kg)	0.05%	TCVN 6701:2000 (ASTM D2622)
2	Cetane index	46	ASTM D4737
3	90% v/v recovered	360	TCVN 2698:2002 (ASTM D86)
4	Flash point (°C)	55	TCVN 6608:2000 (ASTM D3828)
5	Viscosity at 40 °C (cSt)	3	TCVN 3171:2003 (ASTM D445)
6	Carbon residue (10% btms)	0.3	TCVN 6324:1997 (ASTM D189)
7	Ash (% weight)	0.01	TCVN 2690:1995 (ASTM D 482)
8	Water content (mg/kg)	200	ASTM E203
9	Particulate contaminant (mg/l)	10	ASTM D2276
10	Copper strip corrosion at 50 °C, 3 h	Type 1	TCVN 2694:2000 (ASTM D130)
11	Density at 15 °C (kg/m^3)	850	TCVN 6594:2000 (ASTM D1298)
12	Lubricity, corrected wear scar diameter (µm)	460	ASTM D 6079

Table 3 Engine characteristics of some popular diesel engines in Vietnam

Engine model	Type	Cylinder volume (l)	Compression ratio	Power (kW)	Power density (kW/l of cylinder volume)
D1146	Non-turbocharger	8.1	18	130	16
D1146 TI	Turbocharger	8.1	16.7	150	18.5
DE08 TIS	Turbocharger	8.1	18.5	175.92	21.7
DE12	Non-turbocharger	11.05	17.1	169	15.2
DE12 TIS	Turbocharger	11.05	17.0	227	20.54
Mercedes Benz-180	Non-turbocharger	3.0	17.8	158.2	17.8
Kia bongo III	Turbocharger	3.0	18	67.5	22.5
Test engine	Non-turbocharger	**4.75**	**16**	**60**	**12.5**

the intake pressure after turbocharging for the engine; consequently, the engine power depends only on the charge air density, as shown in Eq. (2)

$$\frac{N_{\text{turbo}}}{N_{\text{non-turbo}}} = \frac{\rho_{\text{turbo}}}{\rho_{\text{non-turbo}}} = \frac{110}{60} \tag{2}$$

We assume that the exchange energy inside the compressor is adiabatic. Therefore, the pressure ratio can be determined by Eq. (3):

$$\begin{aligned}\frac{\rho_{\text{turbo}}}{\rho_{\text{non-turbo}}} &= \left(\frac{P_{\text{turbo}}}{P_{\text{non-turbo}}}\right)^{\frac{1}{k}} \\ &= (\pi_k)^{\frac{1}{k}} \Rightarrow \pi_k \\ &= \left(\frac{\rho_1}{\rho_0}\right)^k = \left(\frac{110}{60}\right)^{1,4} \approx 2.336 \end{aligned} \tag{3}$$

We chose a commercial turbocharger based on a pressure ratio of 2.3 to coincide with the engine characteristics and low cost. The important factors that affect the turbocharger performance are the pressure ratio and air mass flow rate through the compressor. Therefore, the turbocharger chosen in this study needs to maintain the pressure ratio and air mass flow rate, as well as maintain the high efficiency.

Corresponding to the pressure ratio of 2.3, the air mass flow rate is determined by Eq. 4:

$$\begin{aligned} G_a &= \frac{N_e.\lambda.L_0.g_e}{1000} \\ &= \frac{110 * 1.2 * 14.7 * 247.97}{1000} \\ &= 485\ \text{(kg/h)} \end{aligned} \tag{4}$$

where G_a is the air mass flow rate, (kg/h), λ is the air excess ratio at full load, $\lambda = 1.2$, N_e is the engine power, $N_e = 110$ kW at 2200 rpm, L_0 is the stoichiometric ratio (A/F), $L_0 = 14.7$ and g_e is the brake specific fuel consumption, $g_e = 250$ g/kWh at 2200 rpm.

Based on the two basic factors, including the air mass flow, pressure ratio and reference material about available turbochargers on the market, for this study, we chose the GT2252 turbocharger with compressor map, as shown in Fig. 1 [25].

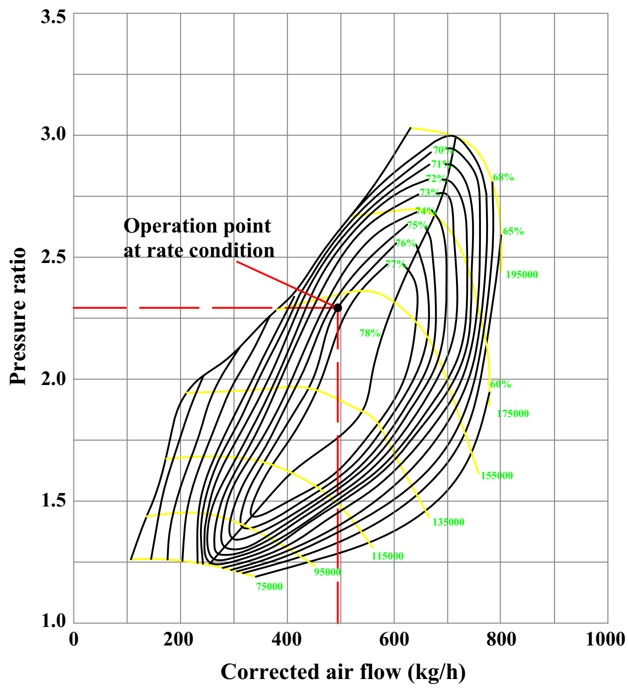


Fig. 1 Compressor map of GT2252 turbocharger

Under rated conditions, the required air mass flow was about 485 kg/h, pressure ratio 2.3 and the operation point decreased in the high efficiency area of the compressor. In addition, the outlet diameter of the compressor and turbo of the GT2252 turbocharger coincided with the intake and exhaust manifold of the test engine; as a result, it was easier to assembly the turbocharger with the test engine.

2.3 Experimental Test of the Engine Performance

We conducted experimental tests to evaluate the performance of the test engine with and without the turbocharger. The tests were carried out for a range of engine speeds ranging from 1000 to 2200 rpm with 200 rpm increments under full-load conditions. The operation parameters of test engines such as the brake power and specific fuel consumption were recorded for evaluation. A schematic of the experimental system is shown in Fig. 2.

The testing equipment includes a chassis dynamometer and a fuel consumption measurement device. All of the devices are controlled directly from a computer using dedicated software, and connected synchronously to a common local network. The chassis dynamometer consists of an absorption unit, and includes the means to measure the torque and rotational speed. Therefore, the engine speed and power can be controlled and measured at the rated power of 220 kW, and the maximum rated value of the rotational speed is 4500 rpm. The dynamometer was controlled using an installed PUMA computer which receives signals from sensors equipped on the dyno and the test engine. In addition, to ensure accuracy in the experimental test, we employed other equipment such as an external cooling system, AVL 533, which allows continuous temperature control. The quantity of fuel consumed by vehicles is measured by fuel consumption measurement equipment, named Fuel Balance AVL 733S, using the gravimetrical method. The fuel system AVL Fuel Balance enables a high-precision fuel consumption measurement even at low consumption and short measuring times. The recommended measuring range of the equipment is up to 150 kg/h with an accuracy of 0.12%.

3 Simulation of Original Test Engine with and Without Turbocharger

We developed simulation models based on structures of the test engine and selected turbocharger using AVL Boost software. Theoretical backgrounds including the basic equation and calculation models for all components of the model are clearly described in [26].

3.1 Basic Conservation Equations

The governing equations of the simulation model are based on the first law of thermodynamics. The first law of thermodynamics for a high-pressure cycle states that the change in the internal energy in the cylinder is equal to the sum of the piston work, fuel heat input, wall heat losses and the enthalpy flow due to blow-by, as given in Eq. (5):

$$\frac{\mathrm{d}(m_\mathrm{c}.u)}{\mathrm{d}\alpha} = -p_\mathrm{c}.\frac{\mathrm{d}V}{\mathrm{d}\alpha} + \frac{\mathrm{d}Q_\mathrm{F}}{\mathrm{d}\alpha} - \sum \frac{\mathrm{d}Q_\mathrm{W}}{\mathrm{d}\alpha} - h_\mathrm{BB}.\frac{\mathrm{d}m_\mathrm{BB}}{\mathrm{d}\alpha} \quad (5)$$

We can calculate the variation in the mass of the cylinder from the sum of the in-flowing and out-flowing masses:

$$\frac{\mathrm{d}m_\mathrm{c}}{\mathrm{d}\alpha} = \sum \frac{\mathrm{d}m_\mathrm{i}}{\mathrm{d}\alpha} - \sum \frac{\mathrm{d}m_\mathrm{e}}{\mathrm{d}\alpha} - \sum \frac{\mathrm{d}m_\mathrm{BB}}{\mathrm{d}\alpha} + \sum \frac{\mathrm{d}m_\mathrm{ev}}{\mathrm{d}t} \quad (6)$$

where m_c is the mass in the cylinder, u is the specific internal energy, p_c is the cylinder pressure, V is the cylinder volume, Q_F is the fuel energy and Q_W is the wall heat loss. α is the crank angle, h_BB is the enthalpy of blow-by, m_BB is the blow-by mass flow, d_mi is the mass element flowing into the cylinder, d_me is the mass element flowing out of the cylinder, and m_ev is the mass of the evaporating fuel.

3.2 Combustion Model

This simulation used the mixing controlled combustion (MCC) model for the prediction of the combustion character-

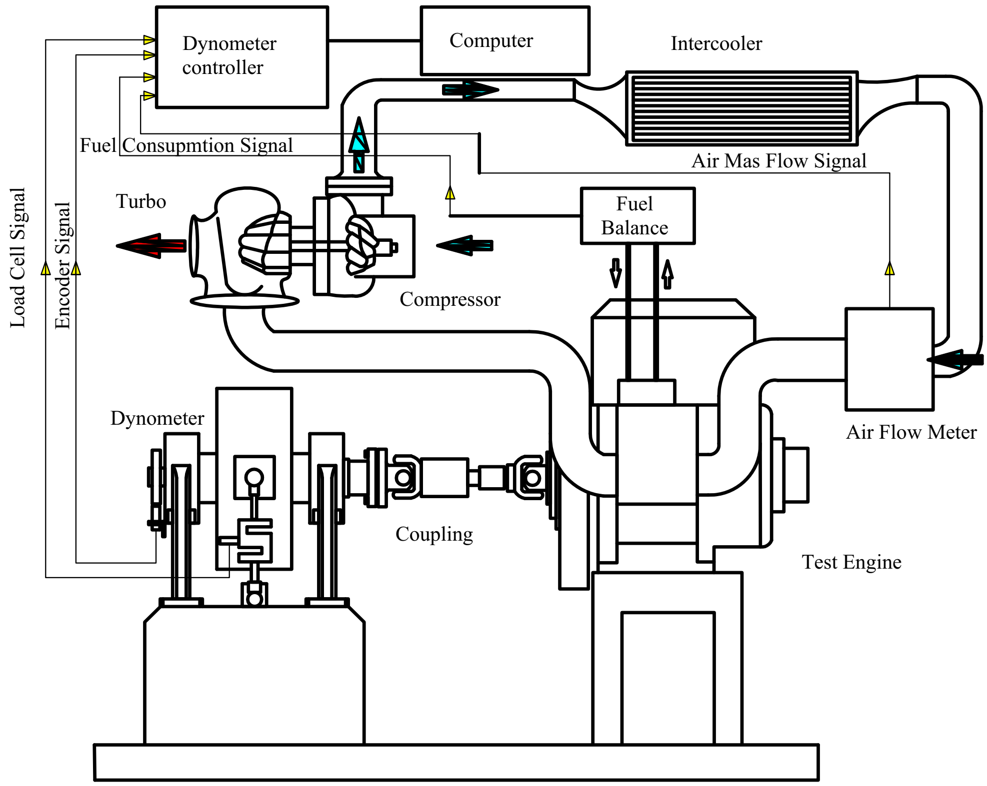


Fig. 2 Schematic of the experimental procedure

istics in direct-injection compression ignition engines. The heat release is a function of the fuel quantity available (f_1) and the turbulent kinetic energy density (f_2), as shown in Eq. (7) [26]:

$$\frac{dQ}{d\phi} = C_{\mathrm{Comb}} \cdot f_1(M_F, Q) \cdot f_2(k, V) \tag{7}$$

where $f_1(M_F, Q) = M_F - \frac{Q}{\mathrm{LVC}}$; $f_2(k, V) = \exp(C_{\mathrm{rate}} \cdot \frac{\sqrt{k}}{\sqrt[3]{V}})$, C_{Comb} is the combustion constant, C_{rate} is the mixing rate constant, k is the local density of turbulent kinetic energy, M_F is the vaporized fuel mass, LCV is the lower heating value, Q is the cumulative heat release for the mixture-controlled combustion, and V is the chamber cylinder volume.

3.3 Heat-Transfer Model

The heat transfer to the walls of the combustion chamber, i.e. the cylinder head, the piston and the cylinder liner, is calculated from Eq. (8)

$$Q_{\mathrm{wi}} = A_{\mathrm{i}} \cdot \alpha_i \cdot (T_{\mathrm{c}} - T_{\mathrm{wi}}) \tag{8}$$

where Q_{wi} is the wall heat flow, A_{i} is the surface area, α_{i} is the heat-transfer coefficient, T_{c} is the gas temperature in the cylinder, and T_{wi} is the wall temperature.

The heat-transfer coefficient (α_{i}) is usually calculated using the Woschni model, which was published in 1978 for high-pressure cycles, and which is summarized as follows:

$$\alpha_{\mathrm{w}} = 130.D^{-0.2}.p_{\mathrm{c}}^{0,8}.T_{\mathrm{c}}^{-0.53}. \times \left[C_1.c_{\mathrm{m}} + C_2.\frac{v_{\mathrm{D}}.T_{\mathrm{c1}}}{p_{\mathrm{c},1}.V_{\mathrm{c},1}}.(p_{\mathrm{c}} - p_{\mathrm{c},0})\right]^{0.8} \tag{9}$$

where $C_1 = 2.28 + 0.308.c_{\mathrm{u}}/c_{\mathrm{m}}$; $C_2 = 0.00324$ for direct-injection engines, D is the cylinder bore, c_{m} is the mean piston speed, c_{u} is the circumferential velocity, V_{D} is the displacement per cylinder, $p_{\mathrm{c},o}$ is the cylinder pressure of the motored engine (bar), $T_{\mathrm{c},1}$ is the temperature in the cylinder at the intake valve closing (IVC), and $p_{\mathrm{c},1}$ is the pressure in the cylinder.

3.4 Turbocharger Simulation

To simulate a turbine, we required the performance characteristics along a line of constant turbine. The power provided by the turbine is determined by the turbine mass flow rate and the enthalpy difference over the turbine. Meanwhile, the power consumption of the turbo compressor depends on the mass flow rates in the compressor and the enthalpy difference over the compressor. The latter is influenced by the pressure ratio, the inlet air temperature and the isentropic efficiency of

Table 4 Some elements of the simulation models

No	Element name	Symbol
1	Intake, exhaust pipe	–
2	Boundary elements	SB
3	Plenum	PL
4	Cylinder	C
5	Restriction	R
6	Measuring point	MP
7	Air cleaner	CL
8	Turbocharger	TC
9	Wastegate	WG
10	Aircooler	CL
11	Aircleaner	CO

the compressor. For steady-state engine operation, the performance of the turbocharger is determined by the energy balance or the first law of thermodynamics. The mean power consumption of the compressor must be equal to the mean power provided by the turbine.

Based on the structure of the test engine characteristics, the models of the test engine with and without the turbocharger were built using AVL Boost software, as shown in Fig. 3. Table 4 shows some elements of the model that correspond to the parameters, which were measured from the real test engine and selected turbocharger.

4 Result and Discussion

4.1 Verification and Validation of Simulation Models

Figure 4 shows that the experimental results for the engine power and fuel consumption curves of the test engine with and without the turbocharger have the same trend as the simulation results, verifying that there is good agreement between the experimental and simulation results. The engine powers obtained experimentally are higher by a maximum of 3.7% compared to those obtained by simulation in the cases of the test engine without the turbocharger and 2.8% in the case of the test engine retrofitted with the turbocharger. On the other hand, the maximum difference in fuel consumption is only about 5.1% when compared with the same operating modes of the simulation and experiment. Slight variations between the simulated and experimental data can be attributed to the assumptions used in the simulation, which cannot be controlled in the experiment. However, the differences are trivial; as a result, the simulation models of the test engine with and without the turbocharger can be used to determine the best solution for performance enhancement research of the old-generation diesel engines applied in the experiment.

4.2 Comparisons of Engine Performance and Fuel Consumption

The results in Fig. 4 also show that retrofitting the suitable turbocharger for an original non-turbocharger diesel engine can improve dramatically its engine power as well as reduce its fuel consumption, as is clearly shown in the literature [27–31]. The maximum engine power increases by 89.5% and the maximum fuel consumption reduces by 17.8% when the test engine is retrofitted with the turbocharger. The increase in the power can be explained by the fact that a turbocharger is basically a pump which is used to force more air, and therefore more oxygen, into the engine. Consequently, more fuel can be supplied into cylinders. This means that more power is generated from the combustion processes inside the cylinders. Therefore, turbochargers can significantly increase the power-to-weight power ratio of internal combustion engines. On the other hand, Fig. 4 also shows that the fuel consumption of the test engine retrofitted with the turbocharger is lower than that of the original test engine because the thermal efficiency of the test engine is improved by the turbocharger, as it increases the volume of air entering it, which lays the foundation for combustion of more fuel.

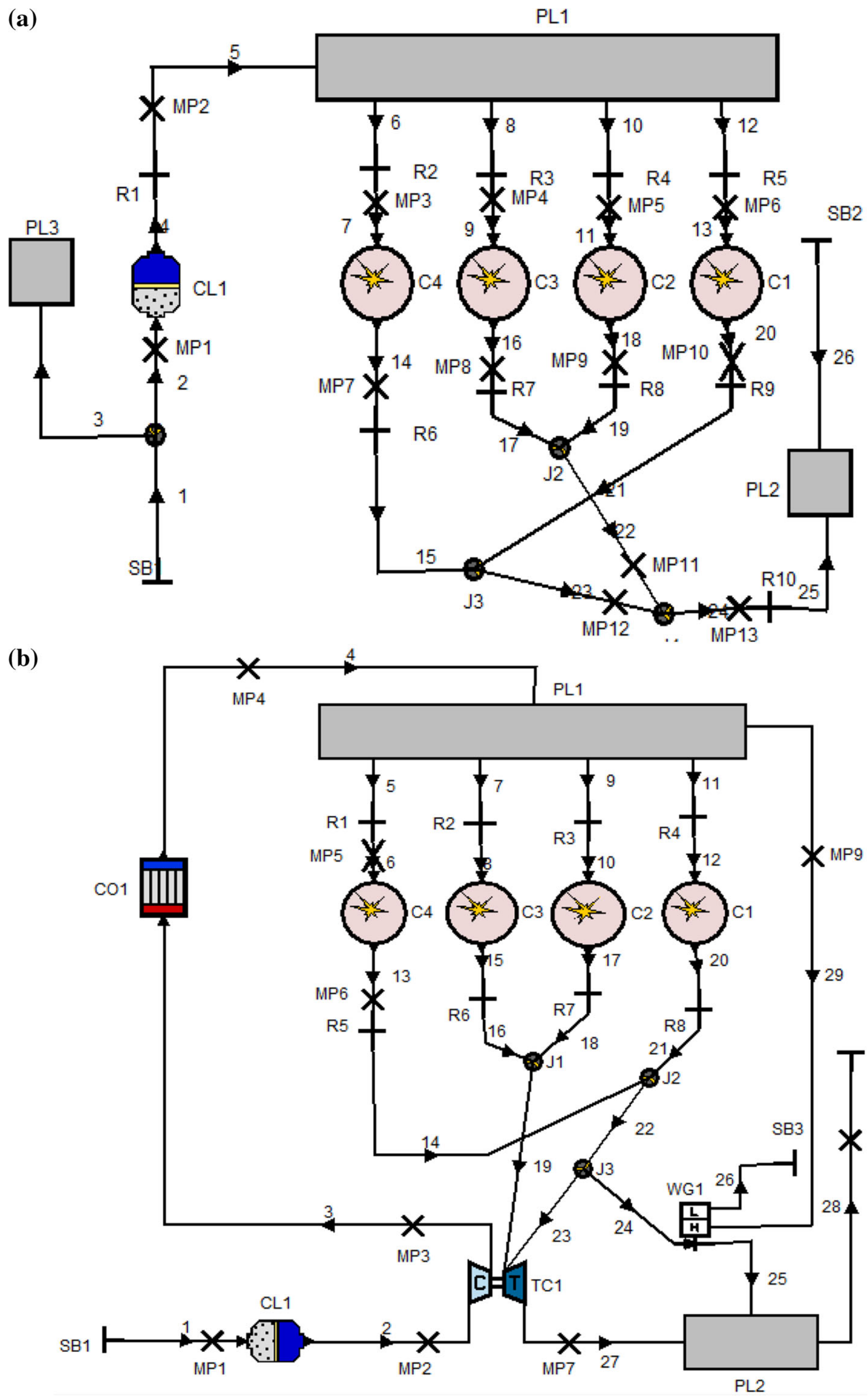
4.3 Comparison of Pressure and Temperature Profiles

Figure 5 shows comparisons of the temperature and pressure profiles of the original test engine and test engine retrofitted with the turbocharger at an engine speed of 2200 rpm. In the case of the turbocharged engine, the in-cylinder pressure and temperature are dramatically higher compared with those of the original engine. These phenomena are in agreement with the results and are well described in many studies [23,31]. The difference can be clearly observed at around the top dead centre, where the rate of burning reached its peak. However, the increase in the pressure and temperature of the turbocharged engine is higher compared to non-turbocharging and will cause an increased load on parts of the internal combustion engine, such as the first ring, piston head, cylinder head and valves. To reduce the mechanical and thermal loads which affect these parts in order to maintain the long-life of the internal combustion engine after turbocharging, several methods should be applied, such as compression ratio reduction, intake-air intercooler, injection adjustment or cooling water flow rate enhancement.

4.4 Comparisons of Combustion Parameters

Figure 6 shows the comparisons of combustion parameters, including rate heat release (HRR), ignition delay and combustion duration of the test engine with and without the

Fig. 3 Models of engines in AVL Boost software: **a** original test engine, **b** test engine retrofitted turbocharger



turbocharger. In the simulation the HRR was modelled as the function of in-cylinder pressure and chamber combustion volume [24,26].

$$\frac{\mathrm{d}Q_\mathrm{n}}{\mathrm{d}t}=\frac{\gamma}{\gamma-1}\cdot p\cdot\frac{\mathrm{d}V}{\mathrm{d}t}+\frac{1}{\gamma-1}\cdot V\cdot\frac{\mathrm{d}p}{\mathrm{d}t} \tag{10}$$

where $\frac{\mathrm{d}Q_\mathrm{n}}{\mathrm{d}t}$ is the net heat release, γ is the ratio of specific heats, p is the in-cylinder pressure, and V is the chamber combustion volume. The in-cylinder pressure was exported from the simulation results corresponding to the last simulation cycle, in which the simulation values are stable (it means the variation values are very small). The peak of the HRR is 124.3 J/° at a crank angle of 370° and 143.0 J/° at

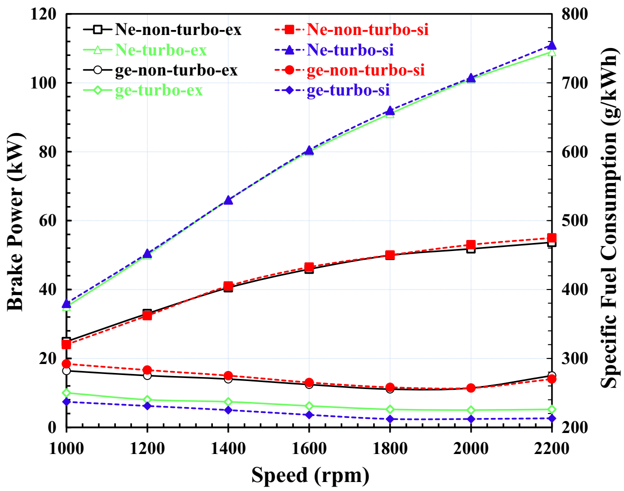


Fig. 4 Comparisons of engine performance and fuel consumption

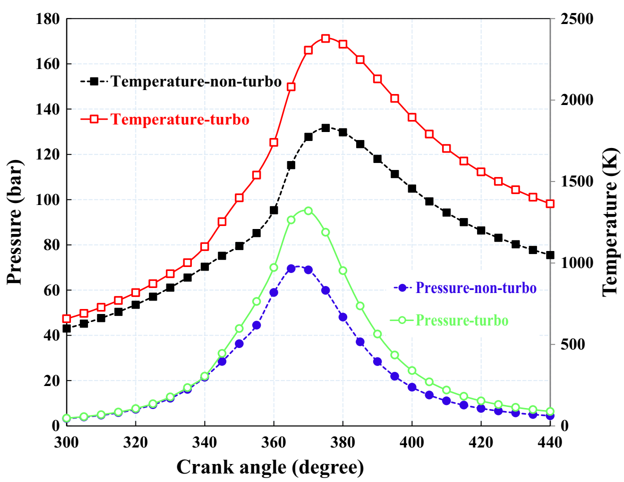


Fig. 5 Comparisons of temperature and pressure at engine speed of 2200 rpm

a crank angle of 365° for test engine without and with the turbocharger, respectively.

Meanwhile, the ignition delay, defined as the time interval between the start of injection and the start of combustion, was modelled as the combination of an Arrhenius (radical production) and Magnussen (influence of turbulence) approach as clearly mentioned in [26]. In a diesel engine, it depends on the fuel characteristic, mixture concentration and its corresponding temperature and pressure conditions. As a result, the higher temperature and pressure on the compression stroke of the test engine retrofitted the turbocharger result in the shorter ignition delay in comparison with that of the original test engine as shown in Fig. 6. Generally, the early ignition delay leads to higher combustion temperatures, although the premixed combustion period is decreased as mentioned in [32]. Therefore, the combustion duration of the test engine retrofitted the turbocharger, defined as the duration of 0–90%

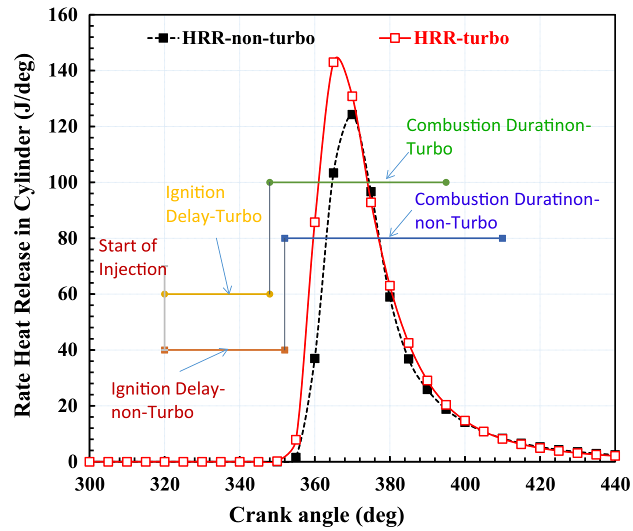


Fig. 6 Comparisons of combustion parameters at engine speed of 2200 rpm

mass fraction burned, is also shorter than that of the original engine as seen in Fig. 6.

4.5 Comparison of Exhaust Emissions

Major emissions from diesel engines include carbon monoxide (CO), nitrogen oxide (NO_x) and particulate matter (PM). CO emissions result when fuel supplied to the engine does not burn completely. Meanwhile, nitrogen oxides, which are collectively known as NO_x, result from the reactions between nitrogen and oxygen atoms under high-pressure and temperature conditions. PM is a combination of other liquid or solid material and soot formed from unburned fuel, which nucleates from the vapour phase to a solid phase in the fuel-rich region of the combustion chamber in an elevated chamber. Consequently, retrofitting the turbocharger for an internal combustion engine will contribute to a reduction in CO, and PM since the fuel burning process is improved owing to the increase in the pressure and temperature, as previously described. However, the NO_x emission of the turbocharged engine is significantly higher than that of the original engine because the formation of NO_x depends on the combustion temperature, as reported in many studies. In this research, we measured and evaluated these gas emissions according to the steady state when controlling the engine speed; as a result, we compared the exhaust-emission components of the engines at various speeds, as shown in Fig. 7. We found that the NO_x emission of the turbocharged engine increased by 11.8% on average compared with that of the non-turbocharged engine. However, we found that CO and PM emissions were approximately 44.7 and 77.0% lower, respectively. The results verify the advantages and disadvantages when retrofitting

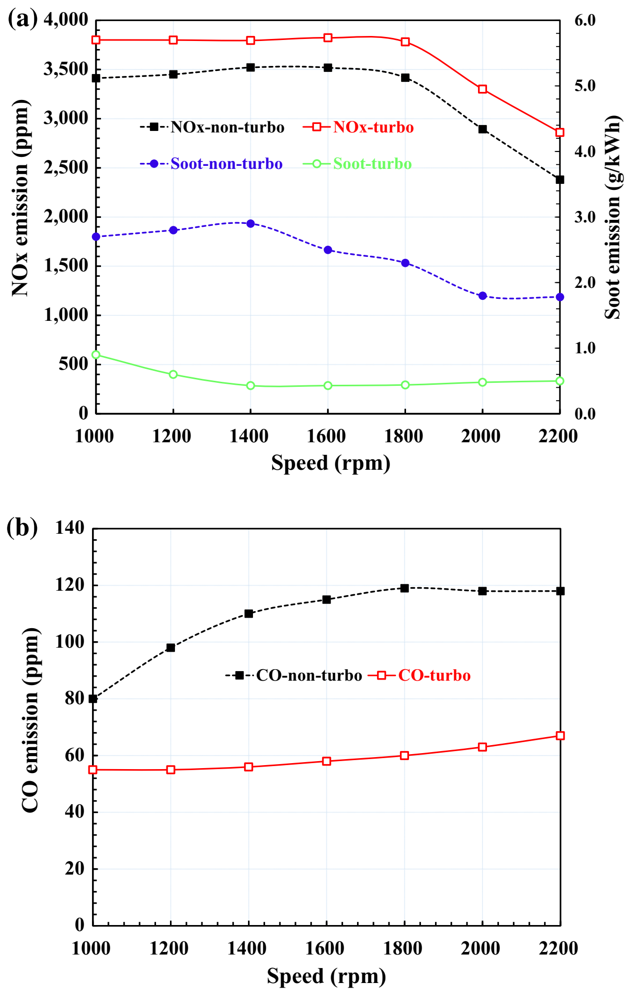


Fig. 7 Comparison of exhaust emissions: **a** NO_x and Soot emission, **b** CO emission

a turbocharger for an internal combustion engine, as clearly described in [23,27–31].

5 Conclusion

In this study, we conducted a simulation and experimental study on the performance characteristics and exhaust emissions of a diesel engine which is not retrofitted and one which is retrofitted with a turbocharger. The results showed that retrofitting the turbocharger for original non-turbocharger engines is a good method of increasing the engine performance as well as reducing HC, CO and PM emissions. This solution can apply for old-generation diesel engines with low power density to improve their effective operations. After turbocharging, the in-cylinder pressure and temperature increase rapidly, which affects the mechanical and temperature strength of parts in the engine such as the crank shaft, connecting rod, piston and cylinder head. In future research, we will continue to optimize our system design including the optimizations of the injection timing and pressure injection in order to enhance the efficiency of the internal combustion engines after turbocharging. In addition, the durability and longevity of the engine parts such as the piston, connecting rob and shaft will be also investigated to fully understand the characteristics of internal combustion engines after turbocharging.

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